IMPROVEMENT OF CENTRIFUGAL COMPRESSOR RELIABILITY HANDLING HIGH PRESSURE AND HIGH DENSITY GAS

by

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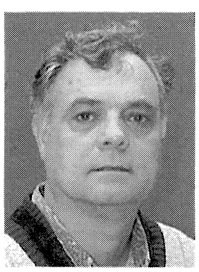

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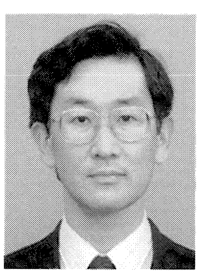

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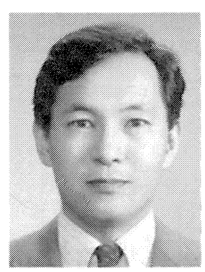

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ABSTRACT

Centrifugal compressors handling high pressure and high density gas can sometimes encounter trouble such as rotor instability and impeller resonance vibration while in operation. In order to eliminate these problems, various analytical and experimental studies were carried out. Shop verification tests were performed to corroborate the studies and to confirm demonstrated improvement in the reliability of the compressors. Reliability improvements in the above two areas—namely compressor rotor stability and impeller resonance—are described in this paper, with specific reference to newly developed knowledge and technology for the benefit of interested centrifugal compressor users and turbomachinery engineers.

INTRODUCTION

Various techniques have been introduced to attenuate rotor vibration of the centrifugal compressor in one of two ways—

namely by increasing the damping force or reducing the exciting force. The former approach includes the application of damper bearings, magnetic bearings, and so on. The latter includes elimination of aerodynamic forces acting on the impellers and reducing the flow-induced force generated through the labyrinth.

Especially in the case of high pressure compressors, because of high density gas, the gas exciting force generated in the impeller and labyrinth is larger than in compressors used for other applications, and is therefore likely to cause unstable rotor vibration. The following new techniques are described and compared with known conventional techniques:

- Installation of an independent damper in the oil film seals.
- Installation of an independent damper at the overhung end of the rotor.
- Control of swirl motion of the labyrinth seal.

Resonance vibration of the impeller, which may lead to stoppage or failure of the compressor, has to be carefully eliminated at the design and manufacturing stage. Advances in finite element calculation methods have enabled a very precise prediction of natural frequencies and mode shapes for complicated impellers. However, the effects of the vibration behavior of the impeller surrounded by high density gas have not been clearly investigated.

A new analytical method for evaluating this effect is described. This method is verified with the impeller vibration data obtained while the compressor is running under full pressure conditions.

ROTOR STABILITY

Rotor stability of turbomachinery is commonly evaluated by comparing the damping ratio and the cross-coupled destabilizing coefficient. Underlying this evaluation method is the basic concept that the final system damping ratio (equals stability) is determined by subtracting the minus components (exciting force) from the plus components (damping force). Figure 1 shows an evaluation block diagram (Kanki, et al., 1988) based on this concept, where each plus and minus component is evaluated quantitatively in terms of stiffness at the midspan of the rotor. (By using the stiffness dimension, each component can be evaluated on the same scale.)

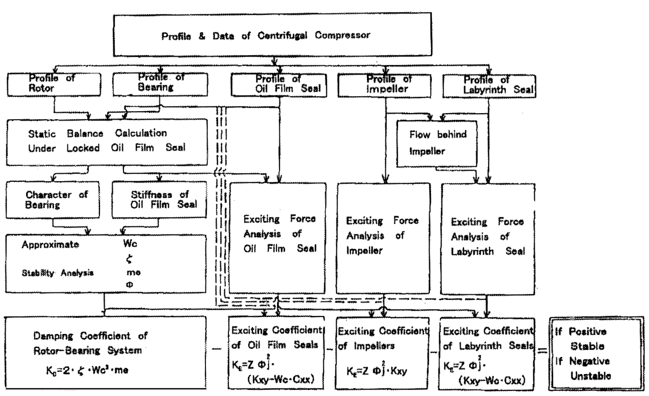


Figure 1. Block Diagram of Stability Analysis Method.

For each plus and minus component shown in the bottom of Figure 1, current quantitative analysis capability varies significantly. The damping coefficient of rotorbearing systems and the exciting coefficient of oil film seals can be rather accurately evaluated, thanks to advances in the rotordynamics calculation code. The exciting coefficient of the labyrinth seals can also be evaluated quantitatively, thanks to available ongoing research work.

On the other hand, detailed investigation for the exciting coefficient of impellers has yet to be done. The formula for predicting this coefficient is provided by the following Wachel equation, modified to Alford force (Wachel, 1975):

$$Kc = \frac{9583.2(kW)(MW)}{D\cdot H\cdot(rpm)} \cdot \frac{\rho d}{\rho s} \qquad (1)$$

where:

kW = Output
MW = Molecular weight of the gas
rpm = Shaft speed
D = Impeller outside diameter
H = Impeller tip opening at discharge
ρ = Density of gas at discharge
ρ_s = Density of gas in suction

With compressors that handle high pressure and high density gas, the outside diameter of the impeller and its tip opening at discharge become very small. As a result, the destabilizing coefficients (i.e., exciting force) of the impellers, Kc, will become larger. Although we do not have sufficient data for supporting this hypothesis, we have to be wary of the increase of destabilizing coefficients.

Therefore, in the case of compressors handling high pressure and high density gas, it is very important to sufficiently increase the damping force of the rotorbearing system and sufficiently decrease the exciting force of the oil film seals and labyrinth seals. In this way, the increasing exciting force of the impellers can be accommodated.

As seen from the above, there are two ways to maintain rotor stability. One method is to increase the damping force of the rotorbearing system, and the other is to decrease the exciting force of the labyrinth seals. Details of these methods are examined below.

The Method of Increasing Damping Force

For compressors handling high pressure and high density gas, possible sources of damping force in the rotorbearing system are bearings or oil film seals. The first approach to bearing seal design was to eliminate the destabilizing force generated by the oil film. The current industrial practice of incorporating tilting pad bearings and multiland seal bushings can successfully ameliorate the destabilizing phenomenon. Additionally, optimization of available damping force can also be performed to some extent—for example, the lower unit load of a bearing generally gives higher damping—however, this approach has its own limitations since dimensional change of these components can adversely affect rotor rigidity or metal temperature.

For these reasons, installation of squeeze film dampers independent of bearing seal design has been proposed. Figure 2 illustrates various positions for, and ways of incorporating the additional squeeze film damping device in a rotorbearing system.

Damper Bearing

Extensive research has been conducted incorporating damper bearings (Zeidan, et al., 1996; Kuzdzal and Hustak, 1996), and damper bearings have been proven as a successful way to increase damping. Damper bearings yield softer bearing and modes (Figure 2(a)), which will improve stability and effective damping. As shown in Figure 1, the damping coefficient of a rotorbearing system is expressed as the quotient of damping ratio, the natural frequency of the basic mode and modal mass. Softer bearings and modes increase the damping ratio and decrease the natural frequency of the basic mode. Thus, careful consideration is required in determining the stiffness and damping behind the bearings so that increases in the damping ratio are not diminished by decreases in the natural frequency of the basic mode.

Nonetheless, one of the drawbacks of the damper bearing is that softer bearings exhibit a higher forced vibration response amplitude (by rotor unbalance or rotating stall) at bearings where vibration monitors are usually installed. Although the actual force transmitted through the bearing is not so different, this higher

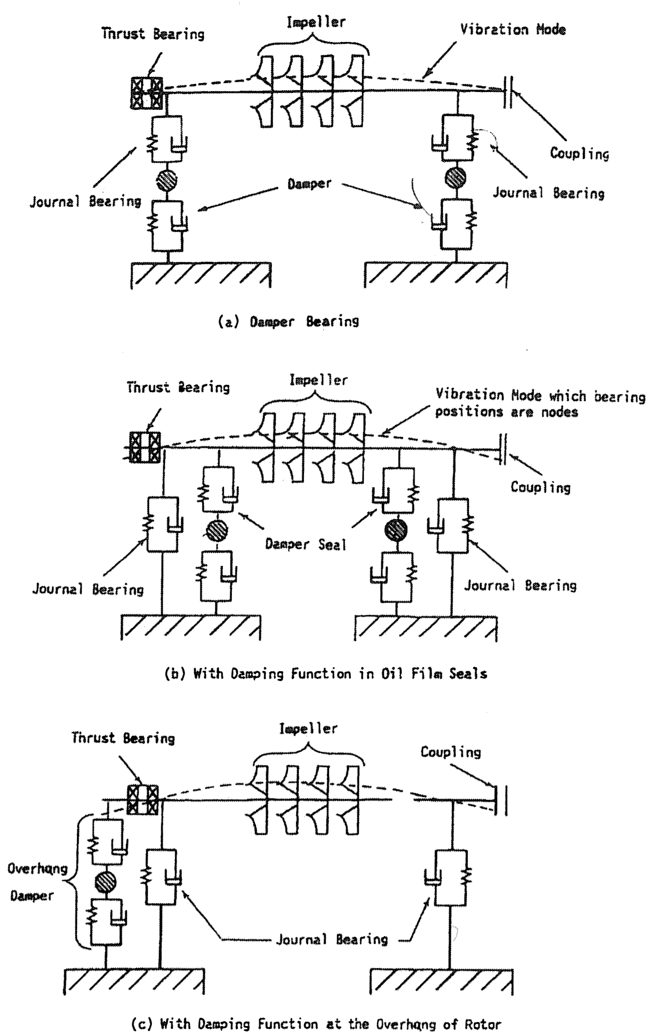


Figure 2. Rotor Models for Rotordynamics Analysis.

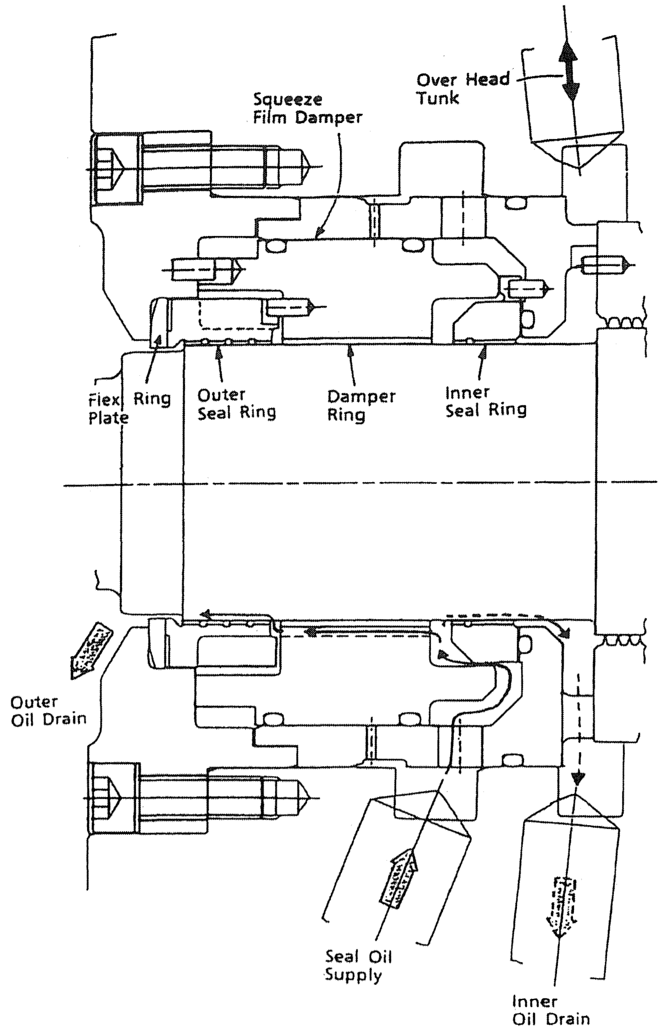


Figure 3. Damper Seal Cross Section.

response amplitude may cause unnecessary concern for those who monitor machine conditions and vibration amplitude levels.

Damper Seal (Damping Function in Oil Film Seals)

When oil film seals are installed on compressors, the seals can be equipped with an oil damping function for rotor vibration by using the seal oil as a squeeze film damper (Figure 2 (b)). The special damping function of this seal (in addition to the ordinary seal function) is achieved by the damper ring, installed between the air side and gas side rings. It follows shaft vibrations with the help of the oil film wedge in the multilobe bearing at the inner periphery, and damps the shaft vibration by squeezing the oil film at the outer periphery. (A typical example is shown in Figure 3.)

As shown by the broken line on a rotor model for rotordynamics analysis (Figure 2 (b)), even if the bearing positions are nodes of the vibration mode, rotor stability is improved by the damping effect of the oil film seals apart from the bearings. The decrease of natural frequency can also be avoided.

Overhung Damper (Damping Function at Overhang of the Rotor)

On the other hand, when dry gas seals are installed on compressors, gases that have far lower viscosity than seal oil (about 1/100) are used for sealing. It is probably impossible to effect damping functions inside gas seals. Thus, new damping functions are effected at the outside of the thrust bearing (Figure 2 (c)) so that even if bearing positions are nodes of the vibration mode, damping effects should be sufficient to ensure no decrease in natural frequency will occur.

The configuration of the new damping functions is similar to an oil film seal with a damping function. If the overhang of the rotor end is too heavy due to the overhung damper mechanism, the mechanism will adversely affect rotor stability. So, the overhung damper has a thin cylindrical shape and is equipped with a thrust disk to ensure sufficient rigidity. (A typical example is shown in Figure 4.)

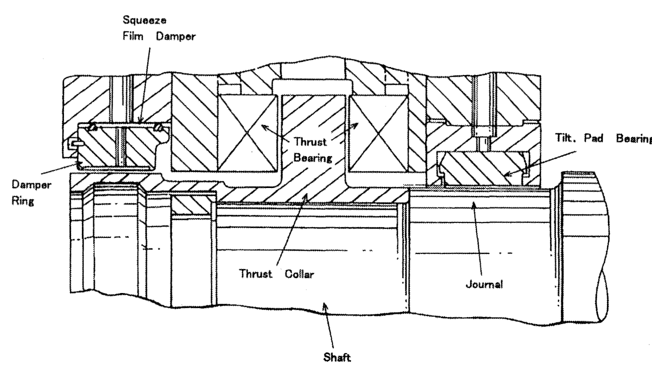


Figure 4. Overhung Damper Cross Section.

Test Rig Data of Damping Effect with Damper Seals and an Overhung Damper

Nonsynchronous sweep excitation tests using the test compressor were performed for the three cases:

- Normal bearing without damping function,
- With damping function in oil film seal (Figure 3),
- With damping function at the overhang of the rotor (Figure 4).

The same rotor and bearings were used in each case. The test rig and test conditions are shown in Table 1 and Figure 5.

Table 1. Test Conditions.

Item	Unit	Value
Bearing span	mm	810
Model rotor weight	kg	150
Bearing size	mm	50
Seal pressure (oil film seal)	barG	20-200
Speed	rpm	0-20,000
Exciting frequency	Hz	0-250

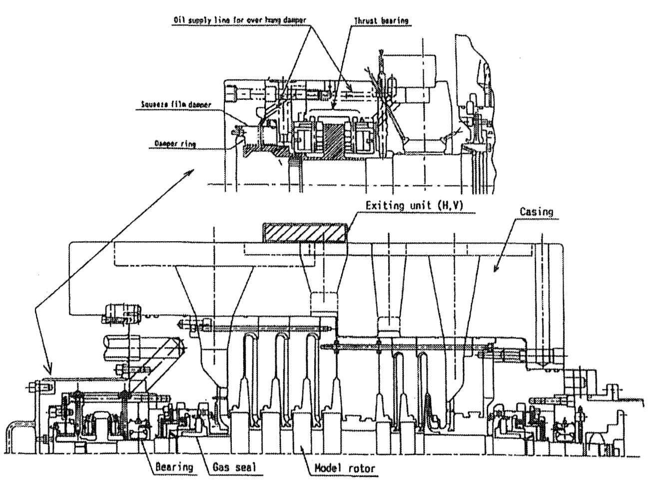


Figure 5. Test Arrangement of Nonsynchronous Sweep Excitation (with Overhung Damper).

Measurement of damping coefficients for each case is performed by nonsynchronous sweep excitation of the whole test rig. Since the rotor stability problem is related to the damping of the nonsynchronous, first natural frequency mode, the best way to measure the location and damping of this mode is by nonsynchronous sweep excitation response measurement (Kanki, et al., 1986). As shown in Figure 6, a 20,000 N hydraulic shaker is mounted on the test rig and rotor vibration response is measured relative to the casing (Figure 7). From this measurement, the increments of damping force are confirmed by the oil film seals or the rotor overhang with damping functions, as shown in Table 2.

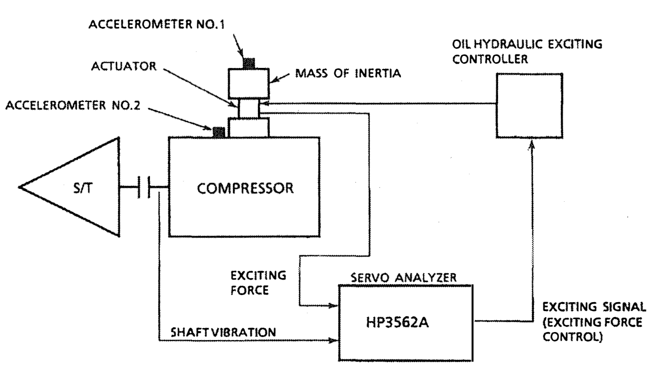


Figure 6. Nonsynchronous Sweep Excitation Test Block Diagram.

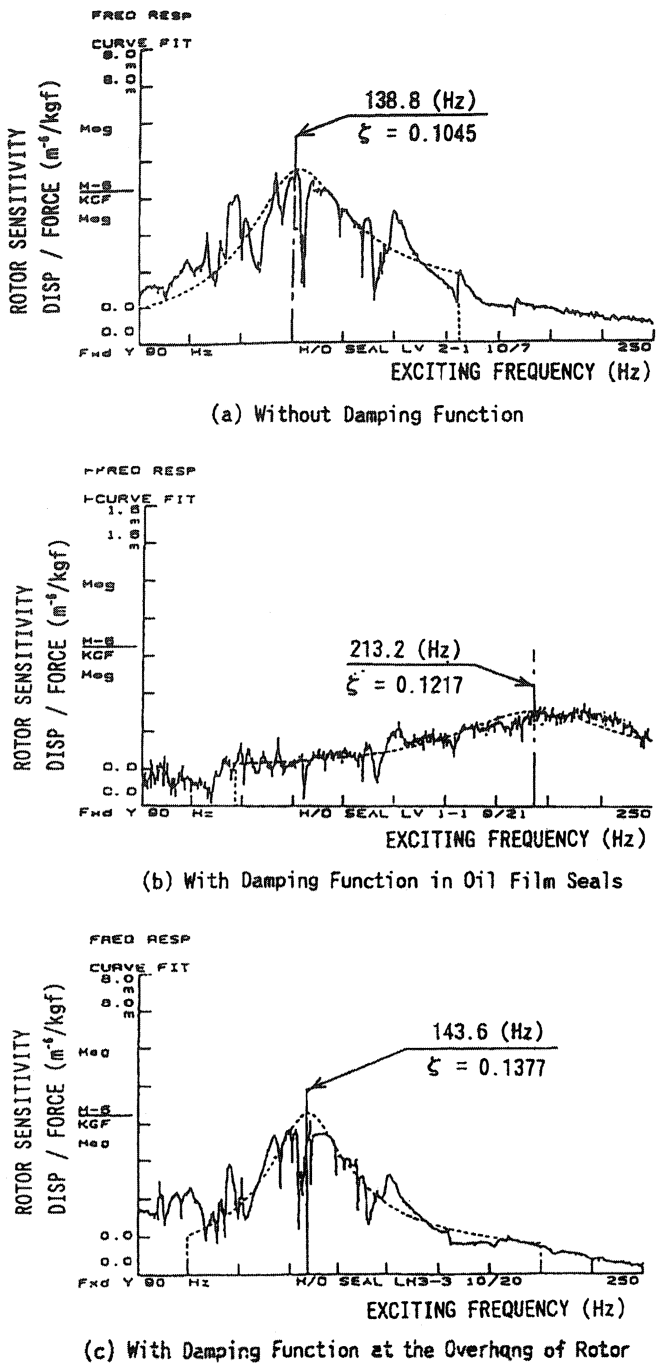


Figure 7. Test Result of Nonsynchronous Sweep Excitation.

Table 2. Nonsynchronous Sweep Excitation Test Results.

	Natural Frequency (Hz)	Damping Ratio (ζ)	Damping Coefficient (Kc)	(Kc/6880) × 100(%)
Without damping function	138.8	0.1045	6880	100
With damping function in oil film seal	213.2	0.1217	18,600	270
With damping function at the rotor overhang	143.6	0.1377	9700	141

The Method of Reducing the Exciting Force

Shunt injection has been the practical approach used to reduce the exciting force of labyrinth seals. High pressure gas is injected into the intermediate labyrinth cavity through the holes located around the circumference of the cavity.

Past research and testing have confirmed that the gas flow at the labyrinth seal—particularly at the inlet swirl—is the major cause of cross-coupled stiffness (the destabilizing force), and that negative swirl can produce negative cross-coupled stiffness (the stabilizing force; Kanki, et al., 1988). Figure 8 shows the typical test results for the small labyrinth seal model (Table 3). To reduce the destabilizing force effect of inlet swirl, two methods of shunt injection were tested: radial bypass and swirl bypass holes. The results show significant increases in the stability limit, and shunt injection allows for stable condition even when inlet pressure is about four times greater than without (Figure 8). As a result, it has been possible to develop a swirl canceling mechanism.

Table 3. Test Model Specifications.

Item	
Seal diameter	100 ϕ mm
Seal radial clearance	0.25 mm
Height of seal fins	2.75 mm
Pitch of seal fins	4.00 mm
Numbers of fins	15 × 2
Inlet pressure	1~3.2 barA
Discharge pressure	1 barA
Critical speed	930 rpm

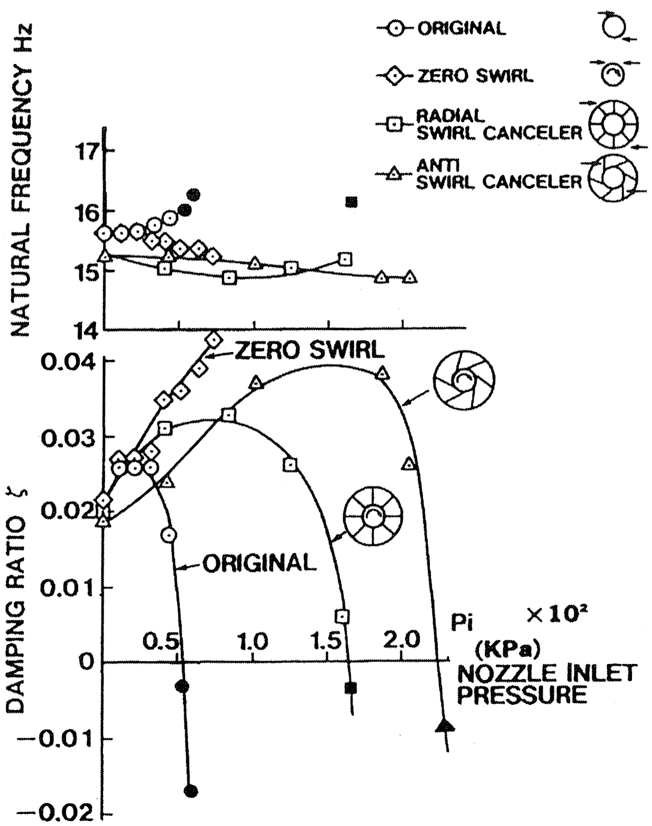


Figure 8. Test Result of Labyrinth Seal Model.

Although all labyrinth seals installed in the compressor inherently possess inlet swirl, it is not practical to apply a shunt injection method for all these labyrinths. The most effective parts for swirl control are balance piston labyrinth and division wall labyrinth, where pressure ratio across the seal is larger than other labyrinths. Accordingly, negative swirl injection is applied for these parts, as illustrated in Figure 9, which produces negative cross-coupled stiffness. Since the rest of the labyrinths without shunt injection produce positive cross-coupled stiffness, total cross-coupled stiffness can be brought quite close to zero.

With this swirl canceling mechanism, the exciting force of labyrinths can be minimized and, at the same time, inner circulating leakage due to shunt injection can be minimized.

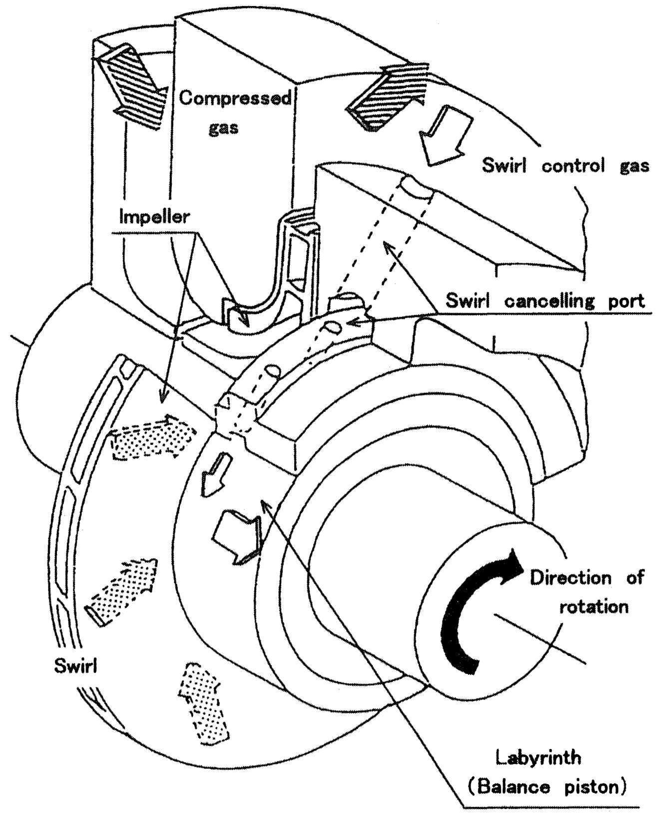


Figure 9. Labyrinth Seal with Shunt Injection.

Test Results of Rotor Stability for Actual Compressors

Shop verification tests of damper seals and overhung dampers were conducted on separate machines before delivery. A damper seal verification test was conducted on the high pressure charge gas compressor for an ethylene plant (Kanki, et al., 1988). This compressor was considered susceptible to the rotor instability problem, as the gas density was sufficiently high to generate appreciable excitation, and the balance piston was located near the rotor midspan. In order to eliminate this problem, gas injection swirl break and damper seals were incorporated.

Full loading was not practical due to the difference of drive turbine steam conditions at the shop. The compressor was half loaded and actual damping was measured by means of nonsynchronous sweep excitation of the whole compressor casing, as in Figure 6. Test results confirmed the credibility of the calculation method and stability prediction (Figure 10).

An overhung damper seal verification test was conducted on the ethylene refrigeration compressor for another ethylene plant. This compressor was also considered susceptible to rotor instability problems, as the bearing span is relatively long for this nearly 10,000 rpm high speed machine with gas seals. The user conducted an independent rotordynamic analysis, including modeling, with and without an overhung damper, and both parties agreed it was best to install the overhung damper for further improvement of rotor stability.

Based on successful testing and operation of damper seals, the verification test was simplified, and a comparison of unbalance

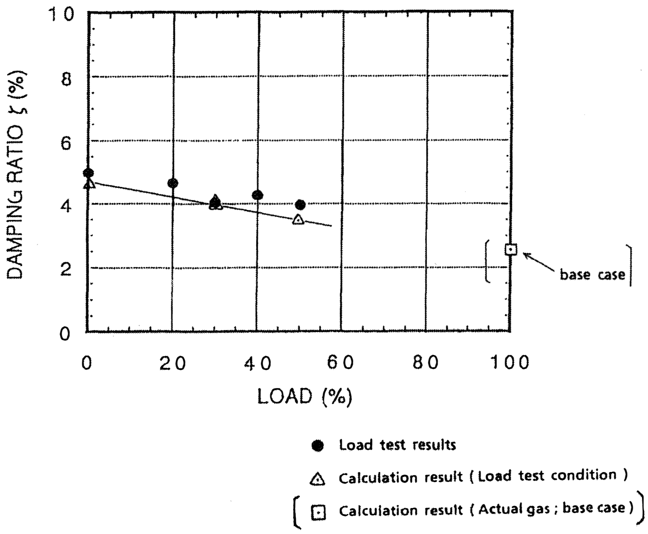


Figure 10. Load Test Result.

response with and without the damper was made during the in-shop mechanical running test. Although dampers are designed to reduce nonsynchronous vibration at normal operating speed, appreciable damping is also expected for the synchronous unbalance response. Test results confirmed the credibility of the calculation methodology and the damping effect of the damper (Figure 11).

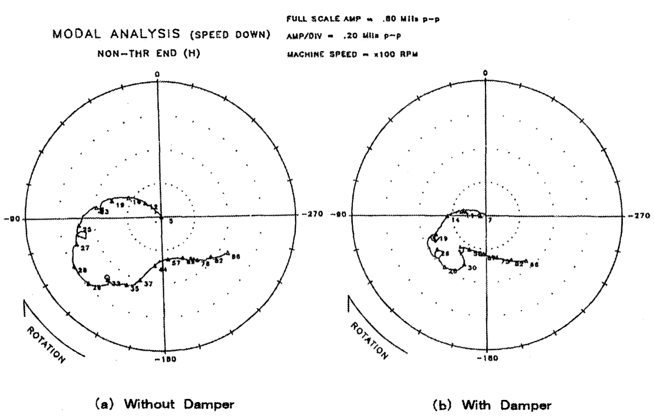


Figure 11. Overhung Damper Verification Test Result.

A USER'S PERSPECTIVE

Some General Background on Instability Concerns

An international petrochemical company keeps detailed records of centrifugal compressor trains worldwide. This includes reliability (forced outages) and availability (planned and forced outages) numbers. These data cover about 100 trains from the late 1970s through today. Historically, rotordynamics problems have been reasonably rare. However, when instability problems have occurred, they have typically been associated with significant business loss. They have typically occurred at higher speeds, with longer rotors, and in higher pressure services. In some cases, application of squeeze film dampers has not been completely successful without some trial and error. Therefore, one has learned to be extra careful when looking at new machines that incorporate these types of additional damping devices. Historical instability problems have been tracked, and these specific data have been used to upgrade rotordynamics analysis computer tools over the years. These historical data are used to better screen new machines for potential rotordynamics problems.

An Independent Rotordynamics Model Is Made

Prior to the purchase of any machine, an up-front design audit is done that includes an independent unbalance response analysis, verification of location of critical speeds, and stability analysis. The vendor is asked, early on, for rotor and bearing geometry so that an independent computer model for rotordynamics analysis can be built.

A Gulf Coast plant expansion project required evaluating new compressor trains with dry gas seals. For this application, the vendor had offered their overhung damper device. These trains would be this company's first field experience with this type of damper arrangement. (Another of this company's plants had already had success with the vendor's oil seal damper arrangement on a machine.) The first step was to model the rotor, assuming the overhung damper was not present. This gives a better feel for the basic robustness of the machine with bearing damping alone. It is preferred that the basic machine have a good log decrement even without the squeeze film damper. The machine should have a big enough shaft to be fundamentally stable with the bearings alone. Then the application of the squeeze film overhung damper would provide additional damping for even more of a stability margin. At the same time, one wants to look at the impact of the damper on the unbalance response results.

Analytical Model Results

Figure 12 shows the screening check used for first pass stability assessment prior to making a detailed model study. It allows one to see where this machine will operate compared with other machines that have had historical instability problems. To date, an instability problem has not been experienced with any machine below the line in the "safe" region. This plot is used for screening machines at the bid conditioning stage. Figure 13 is the vendor's sensitivity analysis of stability margins, both with and without the damper. Both agree that the machine should be stable even without the addition of the damper.

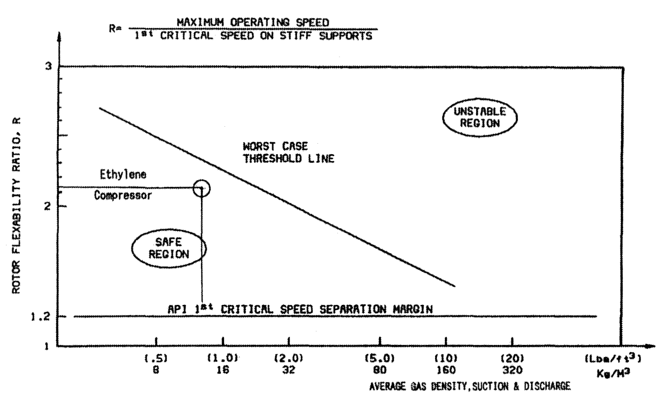


Figure 12. Fulton/Sood Empirical Stability Criteria.

Figure 14 shows the rotor geometry used for detailed, independent rotordynamics analysis. Note that there is only one damper on the free end. Figures 15 and 16 show the tabular results of rotordynamics analysis, both with and without the damper. In all cases shown, the damper adds significant stability margin. In addition, it generally improves the unbalance response at various locations along the rotor. Therefore, the independent stability sensitivity analysis agreed with the vendor's computer analysis. Based on this outcome, the machine's rotordynamics design was accepted subject to a shop sensitivity test with and without the overhung damper, but with applied unbalance to the rotor. (These shop test results are described elsewhere in this paper and shown plotted in Figure 10. Note that the unbalance response on the non thrust end of the machine showed reduced response with the damper, even though the damper is on the opposite end.) Basically,

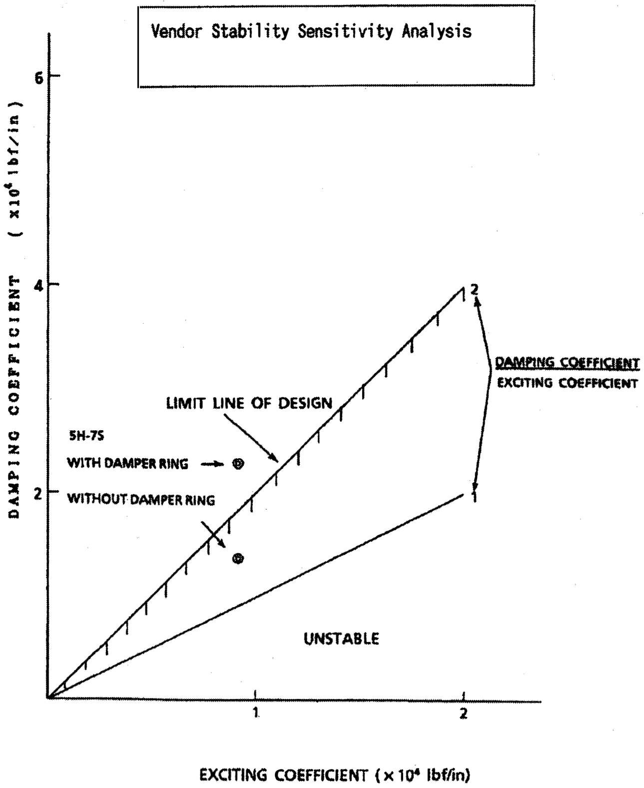


Figure 13. Stability Margin for Nonsynchronous Vibration.

the conclusion is that this machine would run with or without a damper, but that the damper would likely further improve the stability margin. The machine was released for shipment to the field.

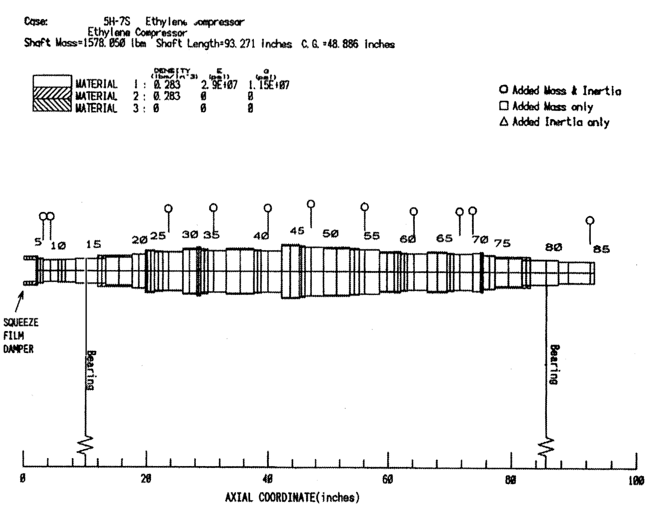


Figure 14. Rotor Geometry.

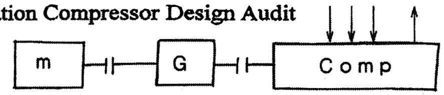
Field Performance Results

The ethylene compressor with the overhung damper design has been running for over three years. Vibrations are low with no evidence of even the slightest subsynchronous component. The same overhung damper device was also used on the propylene and high pressure charge gas rotors with the identical result of trouble-free operations at full capacity rates.

ELIMINATION OF IMPELLER RESONANCE VIBRATION

Resonance vibration of the impeller that may lead to impeller failure has to be carefully eliminated at the design stage. With the

Ethylene Refrigeration Compressor Design Audit



Ethylene Refrig.	
MODEL	5H-7S
ACFM (inlets)	6635/3348/3782
Hp (rpm)	9000 (8576)
Pi/Po <–> Ti/To	15.6/327 psiA <–> -150/102°F

CRITICAL SPEED CHECKS

1st Lateral Critical	3000 - 3500 (2700 - 3400 MHD)
2nd Lateral Critical	11,000 - 12,500 (10,700 - 11,200 MHD)

ZERO Preload; min clr brgs; 5 pad LOP, L/D = .5

	Compressor 8576 rpm: damper-brg-mid-brg	Compressor 1st critical: damper-brg-mid-brg
• Unbal resp, 10X API - mils, PTP		
w/o damper	1.1 .6 1.0 .85	2.2 .9 5.2 .9
with damper	.5 .6 .9 .65	1.3 .9 4.6 .9
• Stability, log dec (no aero)		
w/o damper	.20	.24
with damper	.26	.36
• Stability, log dec (40K aero @ ctr)		
w/o damper	.12	.16
with damper	.18	.28

Figure 15. Rotordynamics Analysis Result (1/2).

ZERO Preload; max clr brgs; 5 pad LOP, L/D = .5

	Compressor 8576 rpm: damper-brg-mid-brg	Compressor 1st critical: damper-brg-mid-brg
• Unbal resp, 10X API - mils, PTP		
w/o damper	2.6 1.5 1.2 1.9	2.0 1.1 5.8 1.0
with damper	.9 .9 .9 .9	1.2 1.1 4.8 1.2
• Stability, log dec (no aero)		
w/o damper	.18	.24
with damper	.23	.33
• Stability, log dec (40K aero @ ctr)		
w/o damper	.16	.21
with damper	.19	.30

Figure 16. Rotordynamics Analysis Result (2/2).

impellers that handle high pressure and high density gas, pressure rise across the impeller becomes higher and, at the same time, the hydrodynamic exciting force acting on the impeller becomes larger. Therefore, it is very important to predict precise natural frequencies of the impeller in operation and to eliminate harmful resonance of impeller natural frequencies with the hydrodynamic exciting force.

For the impeller operating in high density gas (for example, a CO_2 compressor used in fertilizer plants yields a very high gas density at the final discharge section, and specific gravity of gas can be one-third that of water), the effects of virtual mass and the damping effect of the fluid must be accounted for, just the same as for a pump impeller or water turbine runner designed to avoid large errors in estimating the natural frequency, etc. Hence, when designing the compressor impeller, it is important to determine the impeller's structural dimensions by evaluating the vibration characteristics, and to take into account the gas density as well as the stiffness of stationary components such as the diaphragm.

In this approach, the natural frequency of the impeller in high density gas was obtained by applying the natural frequency analysis method employed for water turbine designs (Kanki, et al., 1992). The results showed that the natural frequency can be significantly lowered from that which exists in normal atmosphere. This reduction ratio was factored into the design of the impeller.

The Analysis of Impeller Vibration

Figure 17 shows the virtual mass effect for the ratio of inner and outer diameter about simple bodies (sphere and column). This suggests that when the ratio of inner and outer diameter approaches 1.0, virtual mass is increased by fluid inertia effect.

Namely, if there is a narrow fluid gap around the body and it is microvibrated very fast, the relative movement of fluid becomes very large and false mass is increased by a viscous effect.

Accordingly, we have to estimate the impeller vibration characteristics, taking into consideration the effect of surrounding stationary parts and high density gas in between.

Figure 18 shows, as an example, the condition of the CO_2 compressor impeller. The impeller is modeled with finite elements as illustrated in Figure 19, and a narrow fluid (CO_2) gap between impeller and diaphragm is modeled in the same manner.

As shown in Figure 20, finite element calculation predicted that the natural frequency of the impeller in CO_2 gas would be lowered from 3097.4 Hz to 1971.7 Hz.

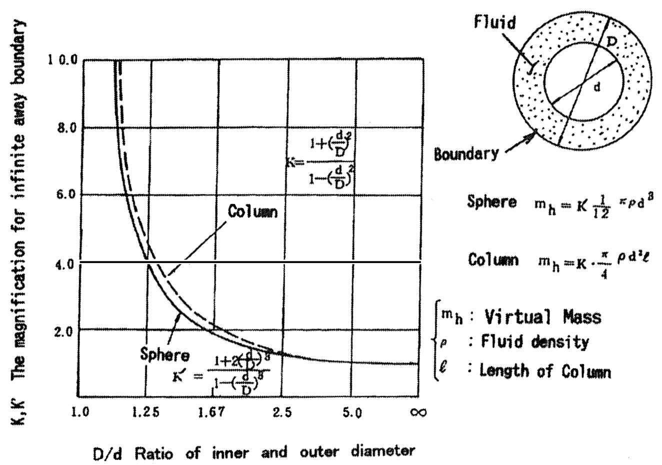


Figure 17. Virtual Mass Effect.

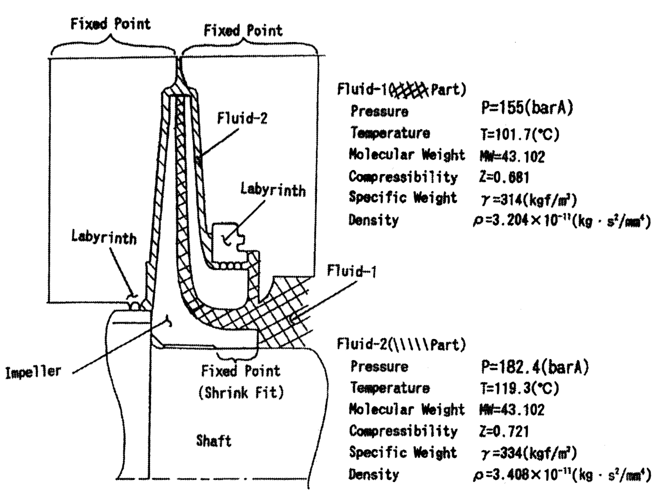


Figure 18. Calculation Condition.

The Test Result of Impeller Vibration

Impeller vibration at the final stage of each section of the CO_2 high pressure compressor was measured using a noncontact displacement sensor, while the machine is run at full pressure condition. Full load condition and sensor arrangement are shown in Table 4 and Figure 21.

The results, as shown in Figure 22(a), show two peaks under full load operation. One was the rotational frequency component "N" and the other was blade running frequency component "NZ."

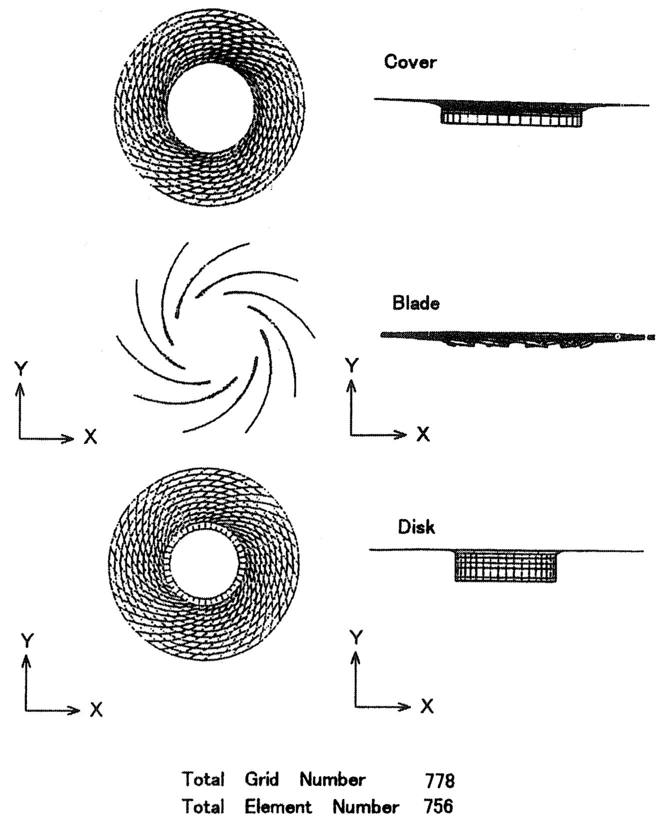


Figure 19. Finite Element Grid.

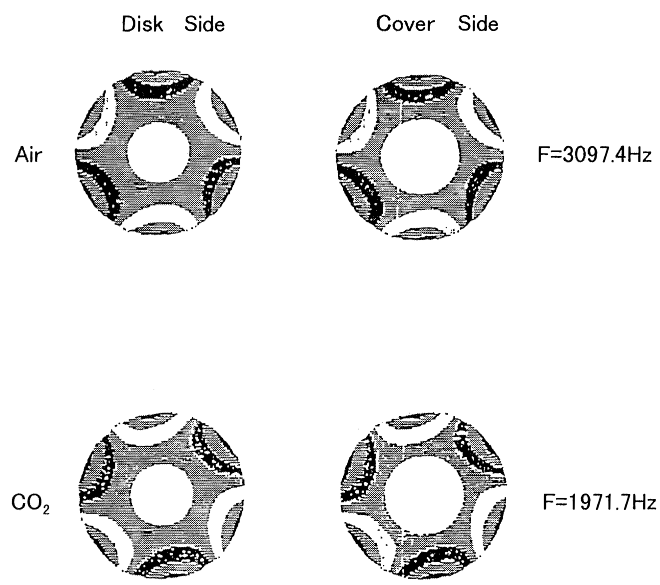


Figure 20. Calculated Vibration Mode (3-ND).

The *N* component was measured due to the impeller angular displacement originating from the rotor assembly work, while the *NZ* component was measured due to the deformation of the impeller, owing mainly to centrifugal force in the low pressure section and gas pressure in the high pressure section, as shown in Figure 22(b). Therefore, there was no sign of abnormal vibration. Natural frequency detected from random vibration is shown in Table 5.

With the reduction ratio of the natural frequency found to be as in the design target, harmful resonance of impeller natural frequencies with hydrodynamic exciting force can be eliminated in

Table 4. Full Load Test Conditions.

Item	Unit	Value
Suction flow rate	m^3/h	400
Suction pressure	BarA	44.1
Suction temperature	°C	40
Discharge pressure	BarA	182.4
Max. continuous speed	rpm	13,490

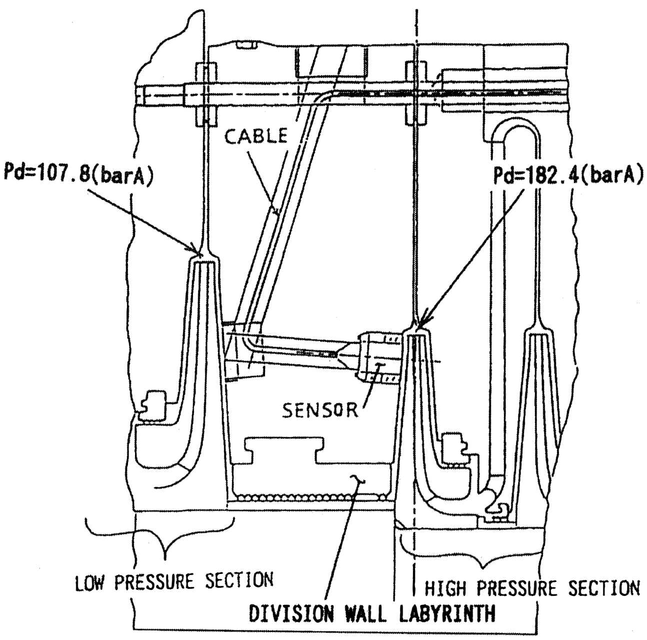


Figure 21. Impeller Vibration Measurement.

the operating speed range, as shown in Figures 23 and 24, which shows the Campbell diagrams both in air and CO_2 gas. In these diagrams, the resonant point is defined by the following equation:

$$| n \times Nz \pm Nd | = H \tag{2}$$

where:

n = Integer
Nz = Number of impeller blades
Nd = Nodal diameter number of impeller vibration mode
H = Possible harmonic hydrodynamic exciting force by return vanes, inlet guide vanes, and diffuser spacers

CONCLUSION

Improvements to the reliability of centrifugal compressors handling high pressure, high density gas conditions have been described. The new damping function increases rotor stability and the swirl canceler decreases exciting force for the rotor.

Furthermore, the impeller resonant vibration can be evaluated by a new analytical method. The authors believe this newly proven technology will be of significant value in improving compressor efficiency and reliability, and plant availability.

REFERENCES

Kanki, H., et al., 1986, "Solving Subsyncronous Vibration Problem by Exciting Test in Operating Condition," Proceedings of International Conference on Rotor Dynamics, IFTOMM, Tokyo, Japan.

Kanki, H., Katayama, K., Morii, S., Mouri, Y., Umemura, S., Ozawa, U., and Oda, T., 1988, "High Stability Design for New

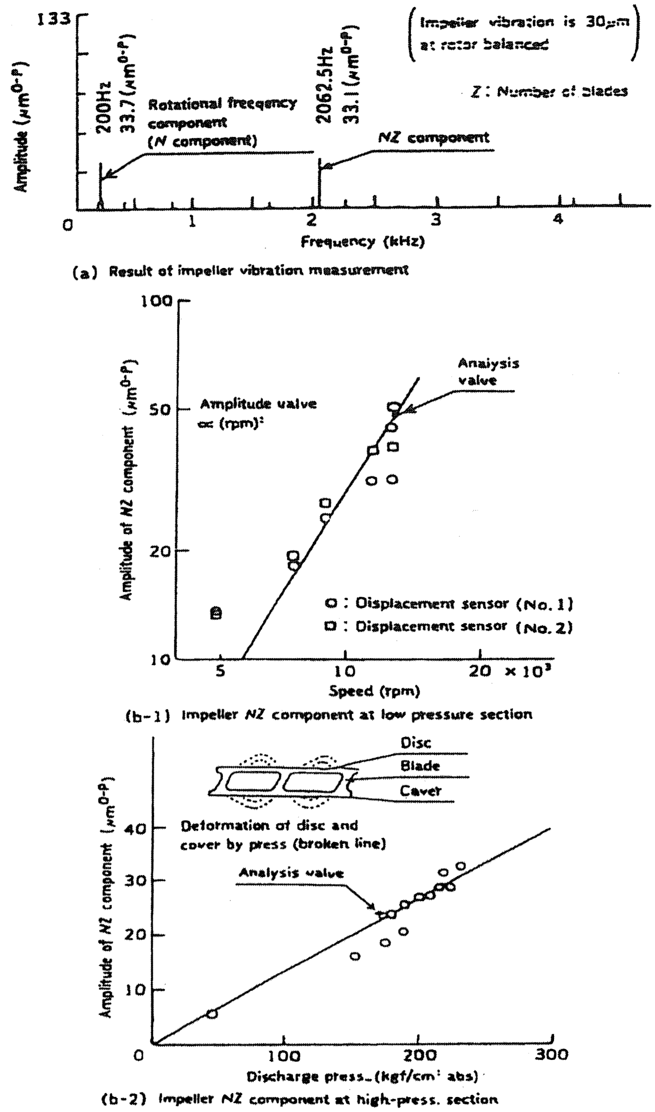


Figure 22. Measured Impeller Vibration Characteristics.

Table 5. Measured Impeller Natural Frequency.

Mode	Final-Stage Impeller		
	Measured Value in Air (A)	Measured Value in CO_2 Gas (B)	Reduction Ratio B/A
0ND	2312 Hz	1492 Hz	0.65
1ND	1960 Hz	1250 Hz	0.64
2ND	2246 Hz	1375 Hz	0.61
3ND	3180 Hz	1995 Hz	0.63

Centrifugal Compressor," Rotordynamic Instability Problems in High Performance Turbomachinery, NASA CP 3026, Proceedings of a Workshop held at Texas A&M University, College Station, Texas, pp. 445-459.

Kanki, H., Mutaguchi, K., Miyagawa, K., Sakamoto, A., Iwasaki, Y., Fujiki, S., Terasaki, A., and Furuya, S., 1992, "Development of Super High Head and Large Capacity Pump-Turbine," Mitsubishi Heavy Industries Technical Review, *29*, (2), p. 116.

Kuzdzal, M. J. and Hustak, J. F., 1996, "Squeeze Film Damper Bearing Experimental Versus Analytical Results for Various Damper Configurations," *Proceedings of the Twenty-Fifth*

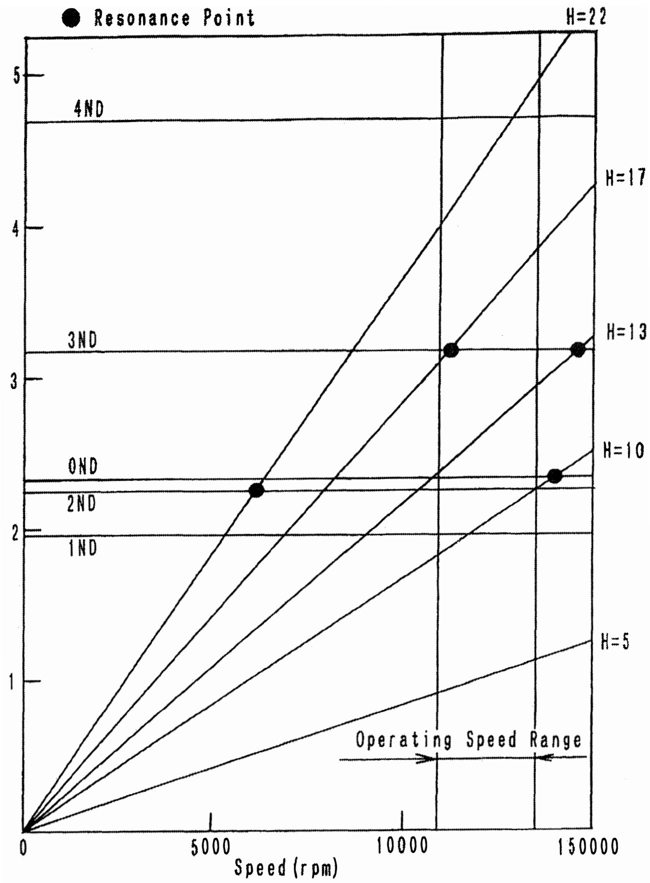


Figure 23. Impeller Campbell Diagram in Air.

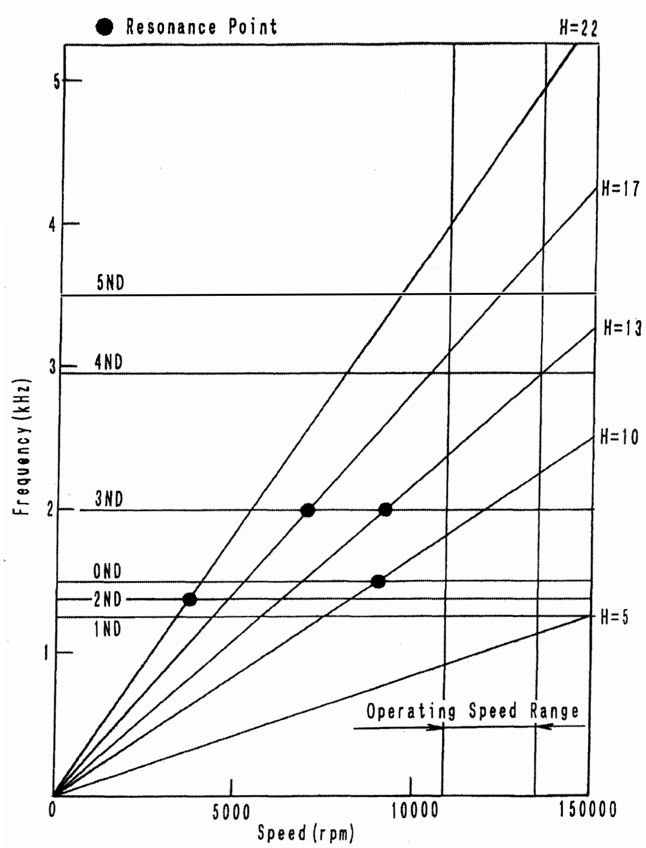


Figure 24. Impeller Campbell Diagram in CO_2.

Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 57-70.

Wachel, J. C., 1975, "Nonsynchronous Instability of Centrifugal Compressors," ASME Paper 75-pet-22.

Zeidan, F. Y., San Andres, L., and Vance, J. M., 1996, "Design and Application of Squeeze Film Dampers in Rotating Machinery," *Proceedings of the Twenty-Fifth Turbomachinery Symposium,* Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 169-188.